NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

WARTIME REPORT

ORIGINALLY ISSUED

September 1945 as
Advance Restricted Report L5G18a

A METHOD FOR DETERMINING OIL-COOLER PERFORMANCE REQUIREMENTS IN SERIES OPERATION

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NACA

WASHINGTON

NACA ARR No. L5G18a [redacted]

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ADVANCE RESTRICTED REPORT

A METHOD FOR DETERMINING OIL-COOLER PERFORMANCE REQUIREMENTS IN SERIES OPERATION

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SUMMARY

An equation has been developed by which the over-all performance of any number of oil coolers when operating in series can be related to their performance when operating singly. This relation is of value to the designer of power-plant installations in determining the individual performance required of identical coolers connected in series to obtain a desired over-all performance. The method is strictly applicable only when there is no oil diversion and when the heat-transfer coefficients on the oil side of the individual coolers do not vary with inlet oil temperature with the coolers in series.

A comparison was made of the experimental over-all performance of two conventional type 15-inch-diameter oil coolers, selected according to the analysis and equipped first with spring-loaded valves and then with rotary valves, with the desired over-all performance. The results showed close agreement between theory and experiment in the region of full oil flow through the cooler cores and in the region where heat-transfer coefficients were not affected by inlet oil temperature and confirmed the anticipated deviations at conditions favorable to oil diversion and variation in heat-transfer coefficients. The analysis made is considered applicable in the range of performance that is important in the design of oil-cooler installations.

INTRODUCTION

The increasing power of aircraft engines has been attended by stricter oil-cooling requirements. New and larger coolers are not always available, however, and in many instances space limitations might prevent their use even if they could be obtained. Multiple oil-cooler installations are therefore used. Multiple oil-cooler arrangements can be classified according to the flow paths of cooling air and oil as either series or parallel for each of the fluids. Series air flow is seldom desirable because of the excessive cooling-air pressure drop and because of the inadequate temperature differentials of the downstream coolers. In the following discussion the air flow is assumed to be parallel, therefore, and the terms "series" and "parallel" will be used to apply only to the oil-flow path.

Series oil flow and parallel oil flow each have their particular advantages and disadvantages, and the choice of arrangement depends upon the needs of the designer. When oil flow from the engine is divided between the coolers, as in parallel arrangement, oil pressure drops are slight. Heat-transfer coefficients, however, are also small - a condition that requires large coolers and that leads to susceptibility to congealing. With the oil flow in series, each cooler receives the full oil flow to the benefit of the heat-transfer coefficients on the oil side but at the expense of oil pressure drop. In parallel operation each cooler performs as an individual unit and, when the oil flow to each cooler is known and assured to be constant, the design problem is essentially no different from that for single cooler installations.

When coolers are connected in series, the over-all performance of several coolers of known performances can be obtained by computing the exit oil temperature from successive coolers. The inverse case of ascertaining the individual performance required of several identical coolers to give a desired over-all performance is usually of greatest interest to the designer. This problem can, of course, be solved by a trial-and-error procedure, which would require much time and effort. The purpose of this report is to develop a formula relating over-all performance in series with individual cooler performances for different coolers and to demonstrate the validity of this formula for two identical coolers equipped first with spring-loaded valves and then with rotary valves. This formula would provide a means of computing the individual cooler performance required to give a specified over-all performance when identical coolers are connected in series.

SYMBOLS

c_p specific heat at constant pressure, Btu per pound per $^{\circ}F$

H heat dissipation, Btu per minute

H' specific heat dissipation, Btu per minute per 100° F temperature difference between average oil and entering cooling-air temperatures $\left(\dfrac{H}{\dfrac{t_i + t_{ex}}{2} - t_{air}} \times 100\right)$

t temperature, $^{\circ}F$

W weight rate of flow of fluid, pounds per minute

η oil-cooler effectiveness on oil side $\left(\dfrac{t_i - t_{ex}}{t_i - t_{air}}\right)$

Subscripts:

air condition of air at entry to oil cooler

1, 2, 3, . . . n first cooler, second cooler, etc.

c common values of a quantity when values at 1, 2, 3, . . . n are equal. For example, when $\eta_1 = \eta_2 = \ldots \eta_n$, then $\eta_1 = \eta_2 = \ldots = \eta_n = \eta_c$

ex condition of oil at cooler exit

i condition of oil at cooler inlet

o over-all

oil condition with reference to oil

t theoretical or specified

exp experimental

ANALYSIS

Measures of Oil-Cooler Performance

Specific heat dissipation.- Specific heat dissipation, defined as the heat transmitted from oil to air each minute when the difference between the average oil and entering cooling-air temperatures is 100° F, is the most commonly used measure of oil-cooler performance. Expressed symbolically, specific heat dissipation is

$$H' \equiv \frac{H}{\frac{t_i + t_{ex}}{2} - t_{air}} \times 100 \qquad (1)$$

where

H heat dissioation, Btu per minute

$\frac{t_i + t_{ex}}{2}$ average temperature of oil in cooler

t_{air} temperature of cooling air entering cooler

Since

$$H = W_{oil} c_{p_{oil}} (t_i - t_{ex})$$

equation (1) can be put in the form

$$H' = \frac{W_{oil} c_{p_{oil}} (t_i - t_{ex})}{\frac{t_i + t_{ex}}{2} - t_{air}} \times 100 \qquad (2)$$

Specific heat dissipation is primarily a function of the weight rates of flow of oil and air and, to a lesser degree, of the temperatures of the incoming oil and air. The nature of the variation of the specific heat dissipation with oil-flow and air-flow rates is illustrated in figure 1 for fixed cooling-air and inlet oil temperatures.

Cooler effectiveness.- Another less frequently used measure of oil-cooler performance is the cooler effectiveness, defined as the ratio of oil temperature drop through the cooler to the difference between inlet oil and entering cooling-air temperatures, or

$$\eta \equiv \frac{t_i - t_{ex}}{t_i - t_{air}} \tag{3}$$

Effectiveness is the degree of completeness with which the maximum available oil-to-air temperature difference is utilized in reducing the temperature of the oil. In the analysis of multiple oil-cooler operation in series, effectiveness is much more amenable to mathematical treatment than is specific heat dissipation. Effectiveness, like specific heat dissipation, is primarily a function of the weight rates of flow of oil and air; the nature of this relationship is illustrated in figure 2.

Relation between specific heat dissipation and effectiveness.- Equation (2) may be rewritten in the forms

$$H' = \frac{2W_{oil}c_{p_{oil}}(t_i - t_{ex}) \times 100}{(t_i - t_{air}) + (t_{ex} - t_{air})}$$

$$= \frac{2W_{oil}c_{p_{oil}} \times 100}{\dfrac{t_i - t_{air}}{t_i - t_{ex}} + \dfrac{t_{ex} - t_{air}}{t_i - t_{ex}}}$$

$$= \frac{2W_{oil}c_{p_{oil}} \times 100}{\dfrac{t_i - t_{air}}{t_i - t_{ex}} + \dfrac{t_i - t_{air}}{t_i - t_{ex}} + \dfrac{t_{ex} - t_i}{t_i - t_{ex}}} \tag{4}$$

By use of the definition of effectiveness as given in equation (3), equation (4) becomes

$$H' = \frac{2W_{oil}c_{p_{oil}} \times 100}{\frac{2}{\eta} - 1} \qquad (5)$$

Equation (5) may also be put in the form

$$\eta = \frac{2}{1 + \frac{2W_{oil}c_{p_{oil}} \times 100}{H'}} \qquad (6)$$

An assumption of 0.5 as a value for the specific heat of oil, which is often of acceptable accuracy, will reduce equation (6) to the form

$$\eta = \frac{2}{1 + \frac{W_{oil} \times 100}{H'}} \qquad (7)$$

Over-All Effectiveness of Several Coolers in Series

In the analysis the following assumptions are made:

(1) There is full oil flow through the cooler cores

(2) Heat-transfer coefficients are independent of the temperature of the oil at the inlet to the cooler

Assumption (2) implies that the effectiveness and specific heat dissipation of successive coolers connected in series are not affected by the decrease in inlet oil temperature accomplished by the preceding cooler.

<u>Effectiveness of n coolers in series</u>.- The overall effectiveness of an oil circuit consisting of n coolers in series is defined as

$$\eta_o = \frac{t_{i_1} - t_{ex_n}}{t_{i_1} - t_{air}} \tag{8}$$

and the expressions for effectiveness of the individual coolers in series are

$$\eta_1 = \frac{t_{i_1} - t_{ex_1}}{t_{i_1} - t_{air}} \tag{9}$$

$$\eta_2 = \frac{t_{i_2} - t_{ex_2}}{t_{i_2} - t_{air}} \tag{10}$$

. . . . .

$$\eta_n = \frac{t_{i_n} - t_{ex_n}}{t_{i_n} - t_{air}} \tag{11}$$

For negligible heat exchanges in the oil line between successive coolers, $t_{ex_1} = t_{i_2}$, $t_{ex_2} = t_{i_3}$, etc., so that equations (10) and (11) become

$$\eta_2 = \frac{t_{ex_1} - t_{ex_2}}{t_{ex_1} - t_{air}} \tag{12}$$

. . . . .

$$\eta_n = \frac{t_{ex_{n-1}} - t_{ex_n}}{t_{ex_{n-1}} - t_{air}} \tag{13}$$

The solutions for exit oil temperature from equations (9), (12), and (13) are

$$t_{ex_1} = t_{i_1} - \eta_1\left(t_{i_1} - t_{air}\right) \tag{14}$$

$$t_{ex_2} = t_{ex_1} - \eta_2\left(t_{ex_1} - t_{air}\right) \tag{15}$$

$$. \quad . \quad . \quad . \quad .$$

$$t_{ex_n} = t_{ex_{n-1}} - \eta_n\left(t_{ex_{n-1}} - t_{air}\right) \tag{16}$$

Successive substitutions of expressions for $t_{ex_{n-1}}$, $t_{ex_{n-2}}$, . . . , t_{ex_2}, and t_{ex_1} as given by equations of the type of equations (14) and (15) into equation (16) yield

$$\begin{aligned} t_{ex_n} = t_{i_1} &\left[\left(1 - \eta_n\right)\left(1 - \eta_{n-1}\right) \cdot \cdot \cdot \left(1 - \eta_1\right)\right] \\ &+ t_{air}\left[\eta_n + \left(1 - \eta_n\right)\eta_{n-1} + \left(1 - \eta_n\right)\left(1 - \eta_{n-1}\right)\eta_{n-2}\right. \\ &\left. + \cdot \cdot \cdot + \left(1 - \eta_n\right)\left(1 - \eta_{n-1}\right) \cdot \cdot \cdot \left(1 - \eta_2\right)\eta_1\right] \end{aligned} \tag{17}$$

The expression for over-all effectiveness, equation (8), thus becomes

$$\eta_o = \frac{t_{i_1}}{t_{i_1} - t_{air}}\left[1 - \left(1 - \eta_n\right)\left(1 - \eta_{n-1}\right) \cdot \cdot \cdot \left(1 - \eta_1\right)\right]$$

$$- \frac{t_{air}}{t_{i_1} - t_{air}}\left[\eta_n + \left(1 - \eta_n\right)\eta_{n-1} + \cdot \cdot \cdot\right.$$

$$\left. + \left(1 - \eta_n\right)\left(1 - \eta_{n-1}\right) \cdot \cdot \cdot \left(1 - \eta_2\right)\eta_1\right] \qquad (18)$$

It can be shown by expansion that the factor multiplying $\frac{t_{i_1}}{t_{i_1} - t_{air}}$ in equation (18) is identical with the factor multiplying $\frac{t_{air}}{t_{i_1} - t_{air}}$. Equation (18) therefore reduces to

$$\eta_o = \frac{t_{i_1} - t_{air}}{t_{i_1} - t_{air}}\left[1 - \left(1 - \eta_n\right)\left(1 - \eta_{n-1}\right) \cdot \cdot \cdot \left(1 - \eta_1\right)\right]$$

$$= 1 - \left(1 - \eta_1\right)\left(1 - \eta_2\right)\left(1 - \eta_3\right) \cdot \cdot \cdot \left(1 - \eta_n\right) \qquad (19)$$

Equation (19) gives the over-all effectiveness of n coolers in series in terms of the effectiveness of the individual coolers. This over-all effectiveness will be called the theoretical over-all effectiveness in contrast with the experimental effectiveness given by equation (8), which is based upon over-all oil temperature drop.

Application of theory to cooler selection for identical coolers.- When n identical oil coolers operate in series and all receive equal amounts of cooling air at the same air temperature, all the units should operate at the same value of effectiveness within the accuracy of the two basic assumptions. This fact is apparent from figure 2, which shows effectiveness as a unique function of oil-flow and air-flow rates. Denoting this common effectiveness by η_c, equation (19) gives for the over-all effectiveness in the special case of identical coolers

$$\eta_o = 1 - (1 - \eta_c)^n \tag{20}$$

By use of the inverse relation

$$\eta_c = 1 - (1 - \eta_o)^{1/n} \tag{21}$$

corresponding to equation (20), it is possible to express the effectiveness required of individual identical coolers, upon which cooler selection depends, in terms of over-all effectiveness desired. The individual effectiveness of each of two identical coolers necessary to achieve a specified over-all effectiveness is given by

$$\eta_c = 1 - (1 - \eta_o)^{1/2} \tag{22}$$

For three identical coolers in series, the necessary individual effectiveness is given by

$$\eta_c = 1 - (1 - \eta_o)^{1/3} \tag{23}$$

Over-All Specific Heat Dissipation of Several Coolers in Series

Although the use of effectiveness was advantageous in deriving the foregoing results, oil-cooler performance is more frequently expressed in terms of specific heat dissipation. The relations obtained by rewriting the results in terms of specific heat dissipation through the use of equation (6) are much more involved, however, than those expressed in terms of effectiveness and are therefore of little practical applicability. Accordingly, in selecting oil coolers for operation in series from charts similar to figure 1, it will be found more convenient to convert the required over-all specific heat dissipation to over-all effectiveness by equation (6), determine the required individual effectiveness by equation (21), and reconvert the required individual effectiveness to the specific heat dissipation by equation (5).

Departure from Basic Assumptions and Effects on the Analysis

Change of heat-transfer coefficient with oil temperature.- Because of the variation of the thermodynamic properties of oil with temperature, the heat-transfer coefficient of oil at constant oil-flow rate diminishes with decreasing oil temperature instead of remaining constant as was assumed in the analysis (assumption (2)). The magnitude of this diminution cannot be predicted exactly because of nonuniformities of local oil-flow rates within the cooler. Furthermore, as the heat-transfer coefficient decreases at low air temperatures, a "waxing" of the heat-transfer surface occurs which reduces specific heat dissipation by rendering parts of the heat-exchanger surface ineffective.

Oil diversion.- The high viscosity of oil at low temperatures creates such a resistance to flow that spring-loaded pressure-relief bypass valves are nesessary to prevent damage to the cooler or other parts of the lubricating system. Such valves are usually closed at normal engine operating conditions, and their action would normally be expected to cause no great deviation of cooler performance from that predicted by the analysis.

In an effort to obtain automatic oil-temperature control and to improve warm-up characteristics, oil coolers are usually fitted with thermostatically actuated oil-flow control valves, which bypass increasing amounts of oil from the heat-exchanger surface as the oil discharge temperature decreases. Such valves are operative whenever the discharge temperature of the oil is below that value for which the valves are set. Since the analysis developed in this report presupposes that the oil flow through each successive cooler in a series is the full amount entering the first cooler and that the entire heat-exchanger surface is utilized at all times (assumption (1)), any action of the thermostatic flow-control valve to bypass oil or eliminate heat-exchanger surface can be expected to cause the performance to deviate from the prediction based on theory. Because the thermostatic valve produces greater diversion of oil as the temperature of the oil decreases, this effect may be expected to become greater with each successive cooler in a series.

If the thermostatic valve is correctly adjusted, however, the entire heat-exchanger surface will be employed at the full oil-flow rate whenever the oil discharge temperature exceeds that for the design condition, if it is assumed that no congealing occurs. Consequently, action of the thermostatically actuated valve can be disregarded in using the analysis for installation design purposes but not in comparing experimental results with calculated performance, except at full oil flow.

APPLICATION AND VERIFICATION OF THE METHOD FOR TWO IDENTICAL COOLERS IN SERIES

The degree to which the applicability of the analysis is affected by actual oil-temperature variation and flow-control-valve operation cannot be determined by pure analysis; therefore comparison with experimentally measured over-all performance is made.

Experimental Data

Individual oil-cooler performance.- Information on the performance of a conventional type 15-inch-diameter oil cooler operating alone at various conditions was obtained from tests conducted at the Naval Air Experimental Station at the Philadelphia Navy Yard and permission to use this information was granted through the courtesy of the Bureau of Aeronautics, Navy Department. The data for an air-flow rate of 318 pounds per minute and an inlet oil temperature of 250° F were reduced to curves of individual effectiveness plotted against oil-flow rate at several cooling-air temperatures to facilitate subsequent computations (fig. 3). The air-flow rate chosen was the one for which there was the widest range of data for oil-flow rate and cooling-air temperature. The solid curves of figure 3 represent performance determined at the Naval Air Experimental Station; dashed parts of the curves represent extrapolations made necessary by incomplete data coverage of the needed operating conditions.

Over-all performance in series.- The actual over-all performance of two previously mentioned conventional type 15-inch-diameter oil coolers in series was also obtained from the Naval Air Experimental Station investigation. The data include operation of coolers equipped with thermostatically actuated rotary flow-control valves and coolers equipped with simple spring-loaded relief valves. Data were selected for the same conditions as those presented for individual cooler performance in figure 3. The data were reduced to curves of over-all effectiveness plotted against oil-flow rate at several cooling-air temperatures and are presented in figure 4. Extrapolation was necessary to obtain outlet oil temperatures at a cooling-air temperature of 100° F; the dashed curves in figure 4 are based on this extrapolation.

Theoretical Over-All Performance in Series

The over-all performance of two conventional type 15-inch-diameter oil coolers in series was calculated by using equation (20) and the values of effectiveness from figure 3 for η_c. Performance so obtained is shown in the form of curves of over-all effectiveness plotted against oil-flow rate at several cooling-air temperatures (fig. 5). Dashed curves are used to indicate the results based on the extrapolated portions of figure 3. It is to be expected that the experimental over-all performance would be the same as that indicated in figure 5, provided the conditions required by the basic assumptions are fully met.

Comparison of Theory and Experiment

The purpose of the analysis is to predict the effectiveness required of each of several identical coolers to obtain a specified over-all effectiveness. In order to verify the validity of the prediction, it is desired to know the agreement between the experimental over-all effectiveness obtained in a series installation of coolers selected according to the analysis and the initially specified over-all effectiveness. For this purpose the theoretical curves of figure 5 can be considered as examples of specified over-all requirements. The curves of figure 4 then represent the experimental over-all effectiveness obtained when the coolers of the required individual effectiveness as given by the curves of figure 3 are connected in series. The curves of figures 4 and 5 can then be compared to demonstrate the validity of the prediction. These comparisons are shown for coolers equipped with spring-loaded valves in figure 6 and for coolers equipped with rotary valves in figure 7.

Spring-loaded valves.- The curves presented in figure 6 show close agreement in most cases between specified and experimental over-all effectiveness. There is, however, an indication of disagreement at a cooling-air temperature of 80° F at very low rates of oil flow. The original data for this test indicate some inconsistency at these conditions, which may be the cause of this discrepancy. Slight disagreement also is shown for a cooling-air temperature of -30° F at very low rates of oil flow. The performance of each of the two coolers operating in series, as obtained from the Naval Air Experimental Station tests, showed that deviation from the performance required of each of the coolers occurred entirely in the downstream cooler. (See fig. 8.) This finding suggests the probability of diminution of heat-transfer coefficient or diversion of oil through the spring-loaded valve as a result of the reduced temperature of the oil entering the second cooler. The very close agreement between specified and experimental over-all effectiveness at an air temperature of 40° F (fig. 9), however, is shown by close agreement between the performance of each cooler in series and the performance required of each cooler. In this case the temperature of the oil entering the downstream cooler was not low enough to alter the performance of the cooler appreciably.

In the range of rates of oil flow and cooling-air temperatures for which the cooler was designed, the maximum percent deviation between specified over-all effectiveness and experimental over-all effectiveness was about 3 percent (fig. 6). This deviation is regarded as well within the accuracy needed for the design of oil-cooler installations.

Rotary valves.- The agreement between the specified and experimental performance curves of figure 7 is generally close at cooling-air temperatures of 100° F, 80° F, and 40° F. Some inconsistency in the original data at air temperatures of 80° F and 40° F was noted at low rates of oil flow and may be the cause of the disagreement at these conditions. At air temperatures of 0° F and -30° F the disagreement was pronounced, except at very high rates of oil flow. The supposition that this condition was caused by a reduction in heat-transfer coefficient and diversion of oil by the bypass action of the rotary valves is substantiated by a comparison of the performance of each of the two coolers in series with the required performance of each of the coolers. Such a comparison of cooler performance for a cooling-air temperature of -30° F is presented in figure 10, which shows how the performance of the downstream cooler was affected by the greatly reduced temperature of the oil entering that cooler.

The maximum deviation between experimental and specified over-all effectiveness was only 5 percent in the range of performance that is important in the design of oil-cooler installations, if some inconsistent data and the extrapolated data for an air temperature of 100° F are disregarded (fig. 7).

General considerations.- In general, for the case of the two conventional type 15-inch-diameter oil coolers connected in series, the agreement between theory and experiment was close in the performance range of importance for oil-cooler installation design. At small oil flows and low cooling-air temperatures, at which the reduction in heat-transfer coefficient is commonly attributed to congealing and the diversion of oil by the flow-control valves is most apt to occur, anticipated deviations were observed. These deviations under congealing conditions are not, however, regarded as detrimental to the applicability of the theory because oil coolers are selected to meet the design heat dissipation at their full oil-flow rate.

CONCLUSIONS

An analysis has been made which is generally applicable to the design of series-flow oil-cooler installations. From a comparison of experimental over-all effectiveness obtained in a series installation of two conventional type 15-inch-diameter oil coolers selected according to the analysis with the initially specified over-all effectiveness, the following conclusions have been drawn:

1. The experimental over-all performance of two identical coolers in series, selected according to the analysis and equipped with spring-loaded valves, agreed closely with the specified over-all performance, the maximum deviation between experimental and specified over-all effectiveness in the performance range of importance for oil-cooler installation design being about 3 percent.

2. Close agreement was obtained in most cases between the experimental over-all performance of two identical coolers in series, selected according to the analysis and equipped with rotary valves, and the specified over-all performance, except in the region where disagreement was anticipated because of the bypass action of the valves. In the performance range of importance for oil-cooler installation design, the maximum deviation between experimental and specified over-all effectiveness was only 5 percent, with the exception of certain data believed to be unreliable.

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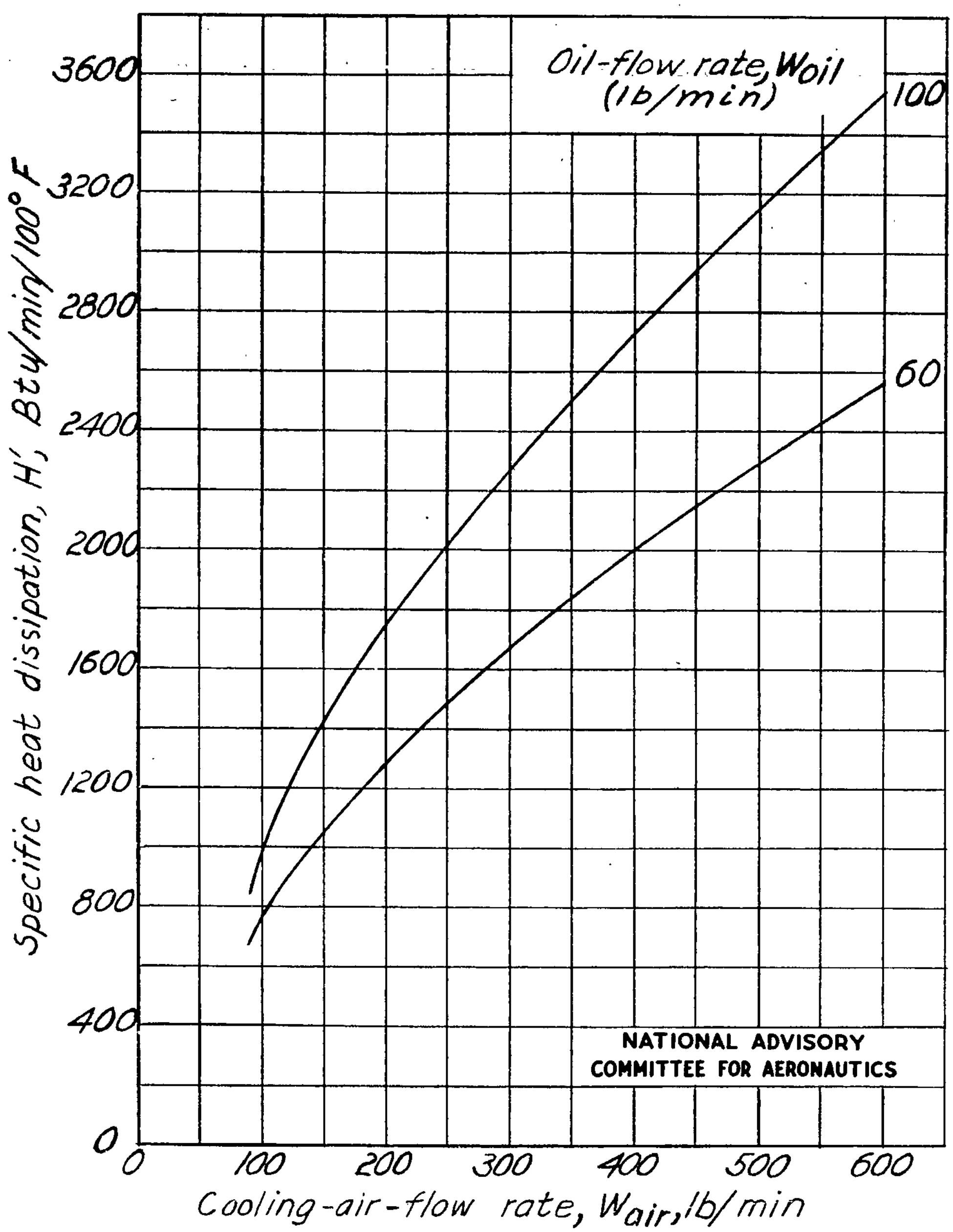


Figure 1.- Example of oil-cooler selection chart based on specific heat dissipation. Inlet oil temperature, 250° F; cooling-air temperature, 100° F.

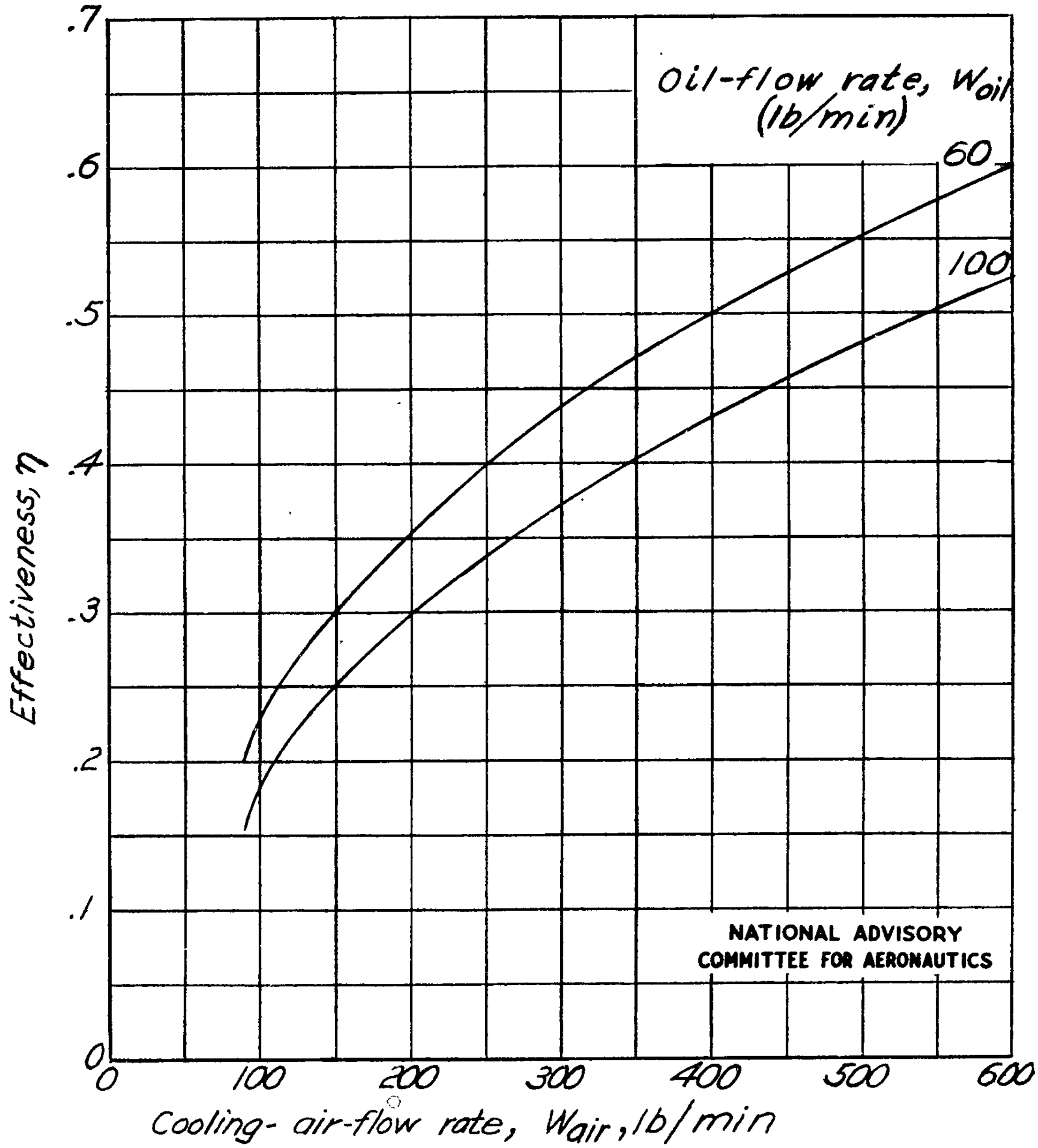


Figure 2.- Example of oil-cooler selection chart based on effectiveness. Inlet oil temperature, 250° F; cooling-air temperature, 100° F.

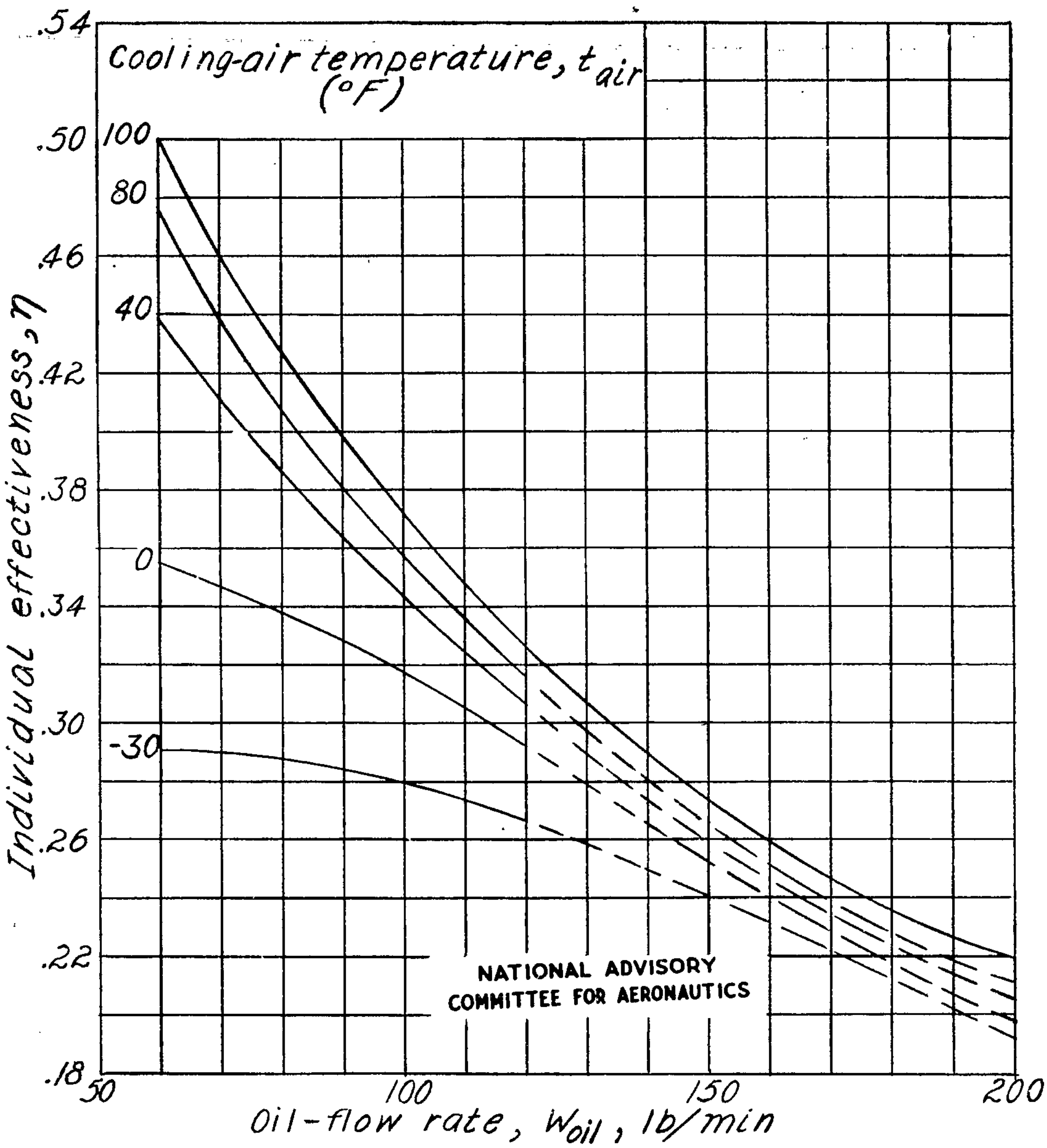


Figure 3.- Individual effectiveness of a conventional type 15-inch-diameter oil cooler. Inlet oil temperature, 250° F; air-flow rate, 318 pounds per minute. (Plotted from data obtained in tests at N.A.E.S., Philadelphia Navy Yard.)

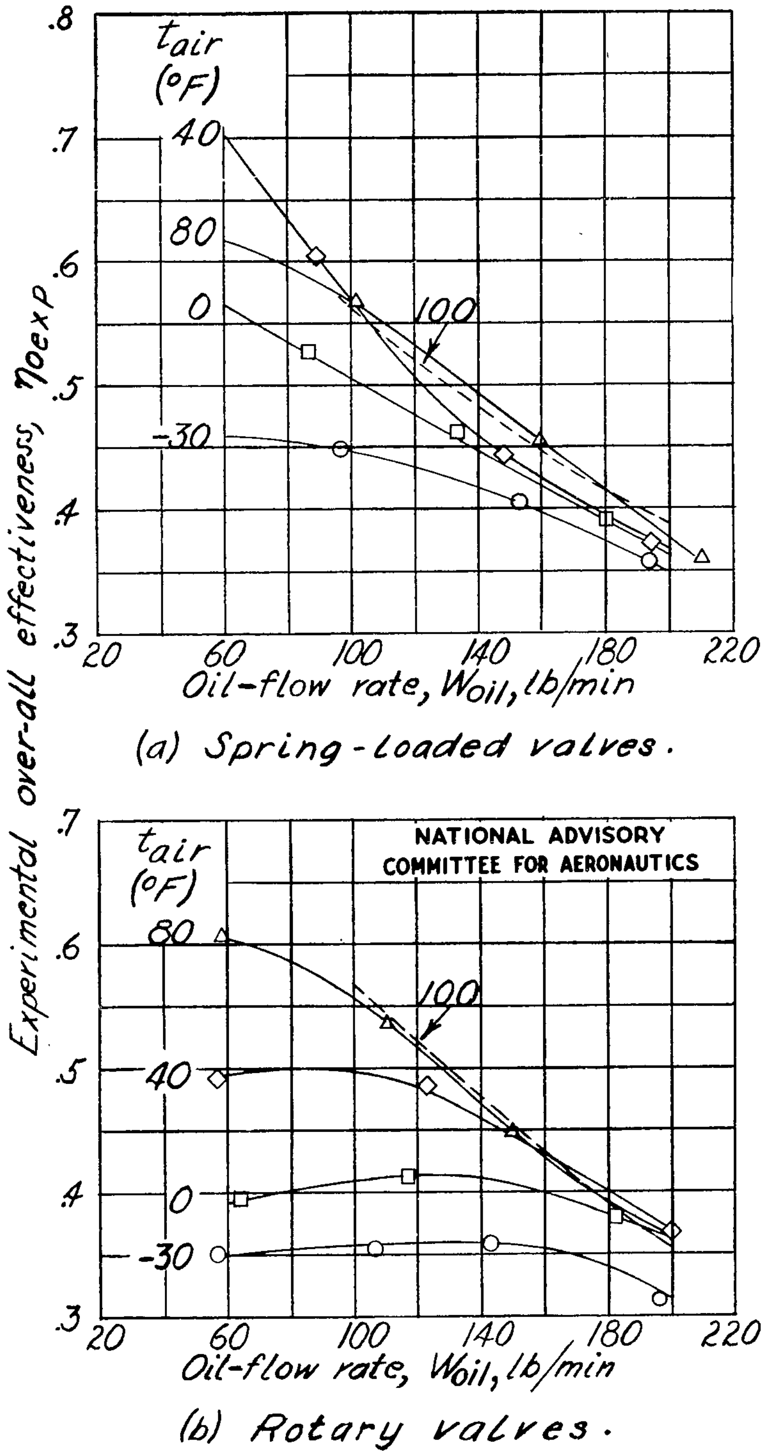


Figure 4.- Experimental over-all effectiveness of two conventional type 15-inch-diameter oil coolers in series. Inlet oil temperature, 250° F; air-flow rate, 318 pounds per minute. (Plotted from data obtained in tests at N.A.E.S., Philadelphia Navy Yard.)

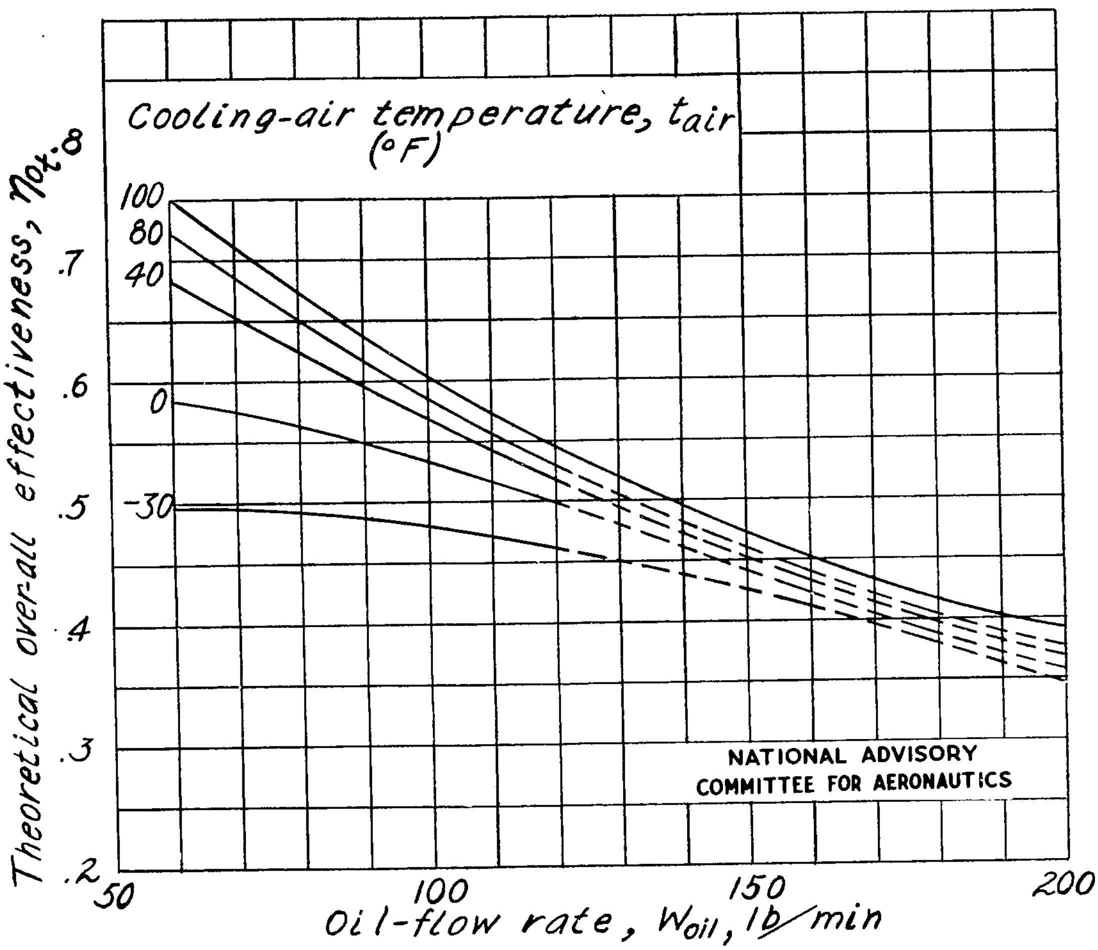


Figure 5.- Theoretical over-all effectiveness of two conventional type 15-inch-diameter oil coolers in series. Inlet oil temperature, 250° F; air-flow rate, 318 pounds per minute.

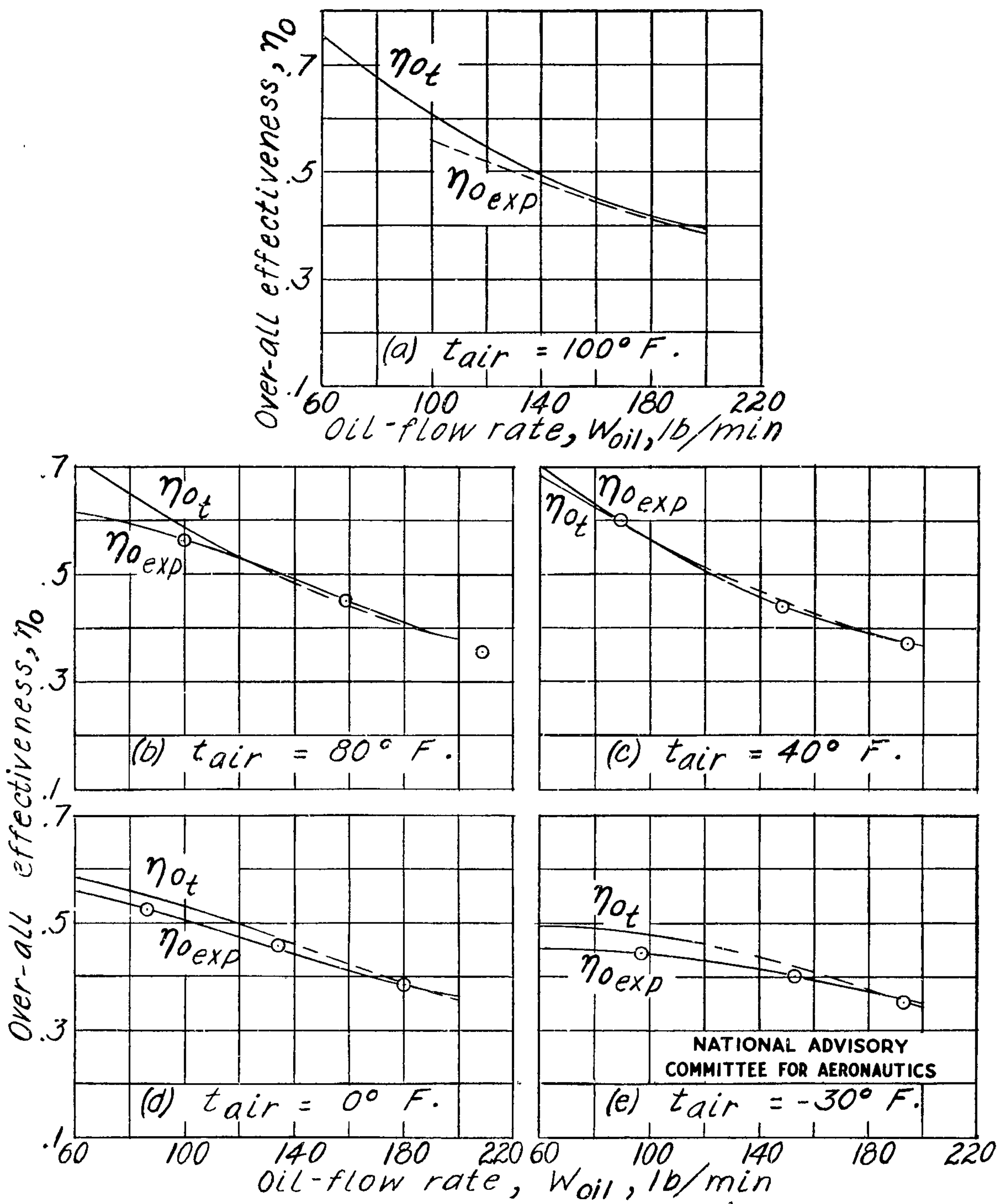


Figure 6.- Comparison of experimental and specified over-all effectiveness. Spring-loaded valves; inlet oil temperature, 250° F; air-flow rate, 318 pounds per minute.

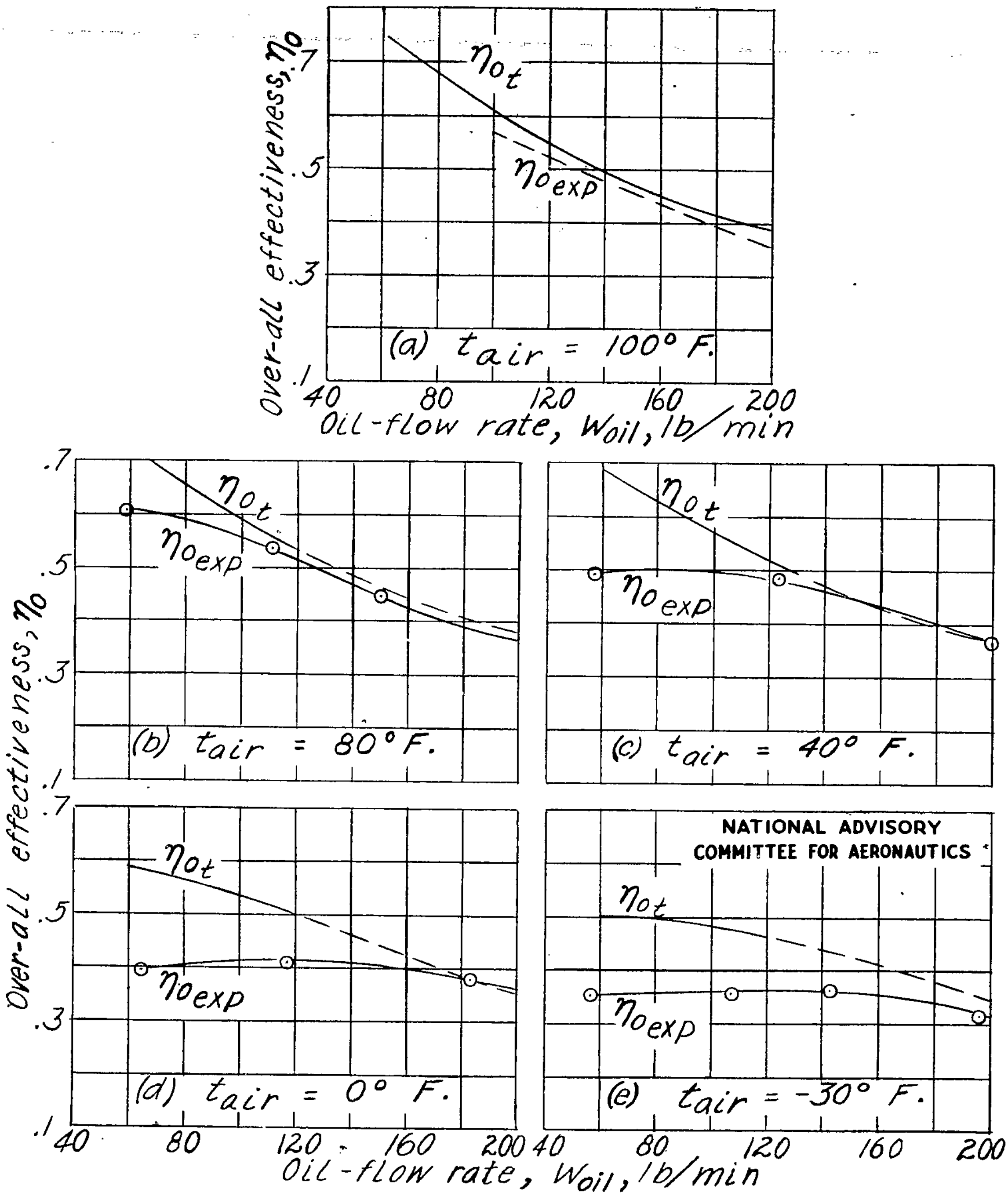
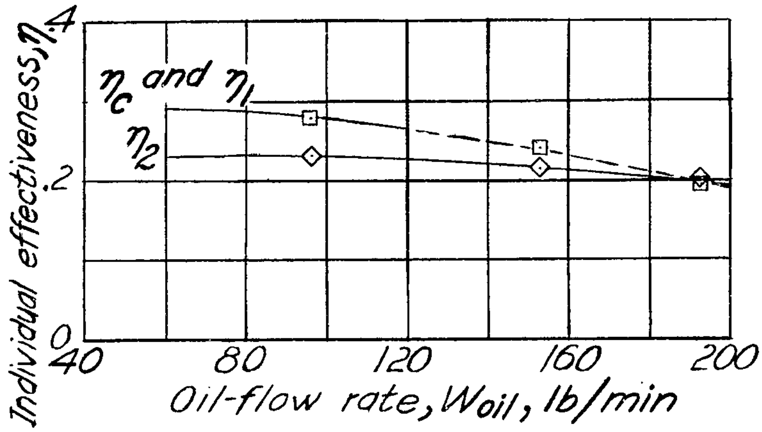


Figure 7.- Comparison of experimental and specified over-all effectiveness. Rotary valves; inlet oil temperature, 250° F; air-flow rate, 318 pounds per minute. Uncorrected for bypass action of rotary valves.



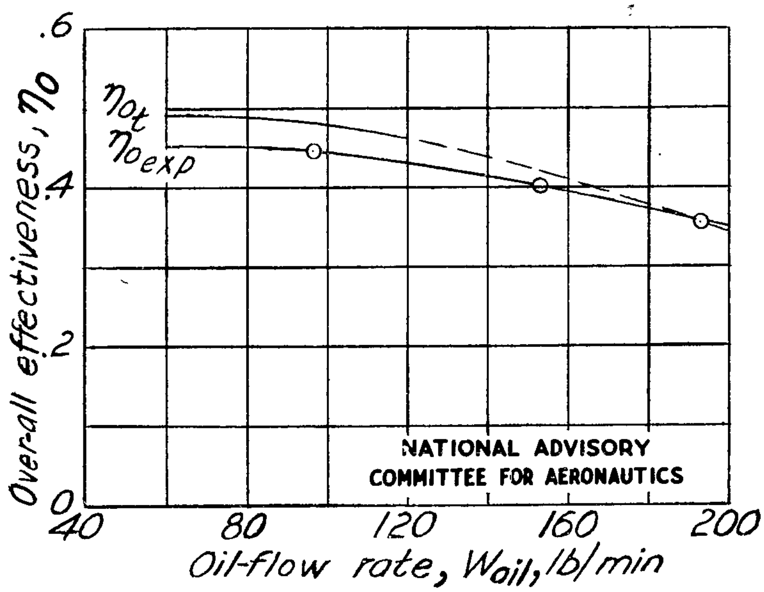


Figure 8.- Comparison of experimental individual and over-all effectiveness with required individual and specified over-all effectiveness, respectively. Spring-loaded valves; inlet oil temperature, 250° F; inlet air temperature, -30° F; air-flow rate, 318 pounds per minute.

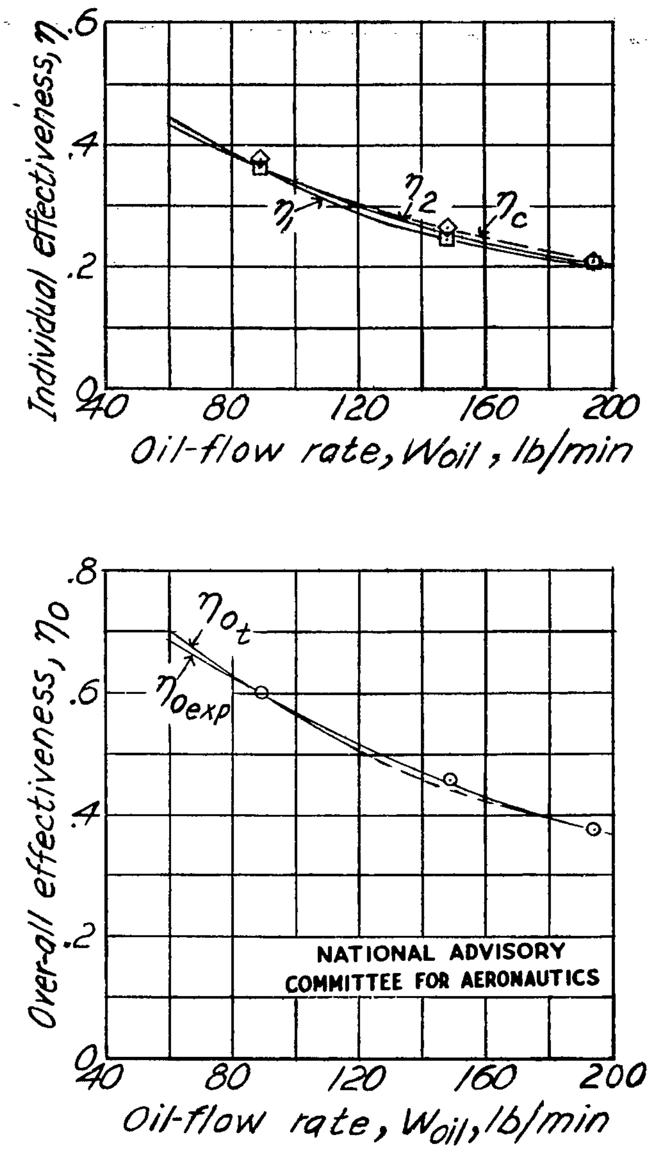
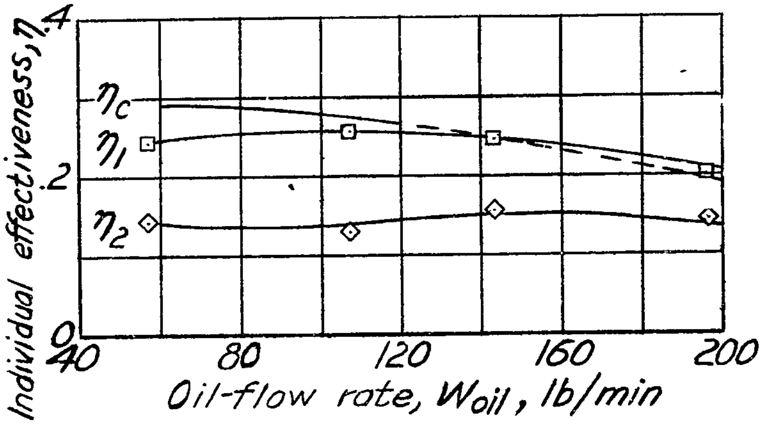


Figure 9.- Comparison of experimental individual and over-all effectiveness with required individual and specified over-all effectiveness, respectively. Spring-loaded valves; inlet oil temperature, 250° F; inlet air temperature, 40° F; air-flow rate, 318 pounds per minute.



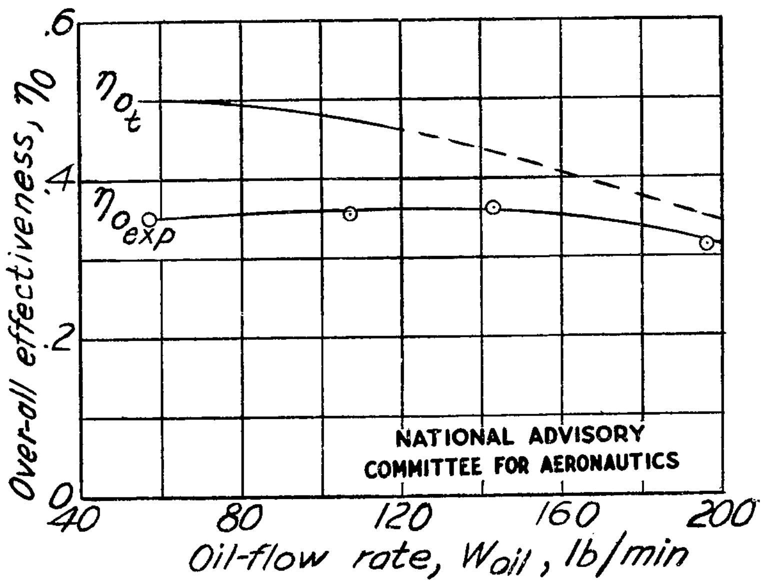


Figure 10.- Comparison of experimental individual and over-all effectiveness with required individual and specified over-all effectiveness, respectively. Rotary valves; inlet oil temperature, 250° F; inlet air temperature, -30° F; air-flow rate, 318 pounds per minute. Uncorrected for bypass action of rotary valves.